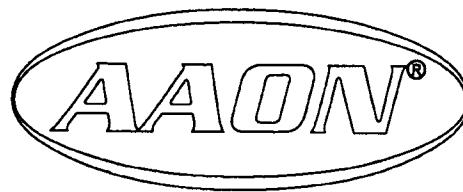


RL Series
45 to 230 tons
Packaged Rooftop Conditioners
& Air Handlers

Engineering Specifications
and Selection Procedures



2425 South Yukon
Tulsa, Oklahoma 74107
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AAON INC

2425 S. YUKON, TULSA, OK 74107 918-593-2286

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PLEASE REMIT TO:
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CUSTOMER NO. **848835**

INV. DATE 02/28/02

PAGE 1

CUST. P. a NO. A-0440

SOLD TO:
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C/O SCALISE INDUSTRIES
601 COLLIER'S WAY
WEIRTON, WV 26082
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Order Form

1025 South Tulsa Ave • Tulsa, OK 74107 • Tel: (918) 661-2255 • Fax: (918) 661-8894		Mon-Fri 8a-4:00 p.m.	
To: Tobby-Karr Sales Agency 4640 Campbells Run Rd Ship To: Philadelphia, Pa. 16806 USA		Date: 11/16/0001 Page 1 of 1 Job Name: Weyton Medical Center Customer P.O. #: 348 Customer #: 348 Shop Order #: 778 Rep #: 1110442 Action: Release To Production Ship: 48 Hours Notify: Joe Snellie 48 Hours Before Delivery @ Phone No: 1-724-748-5400 Marketing Code: Shipping Zone:	
QTY 2		Description RL-156-4-0-AB06-000 : HCC1-K00-QN1-0B0 - JAAC00H-0-0-A000000B TAU UNIT: CTM RTUW 1.3	
Rep Contact: Creed Hess Ordered By: Creed Hess		Unit Net Ext. \$94,880.91	Ext. Price \$189,781.83
		Multiplier: 0.816 Commission %: 0.00	Total Line Price \$: \$152,142.00 Total Net (Ext.) \$: \$189,781.83 Freight \$: \$4,000.00 Commission \$: \$0.00
		Billings: 100%	100% \$189,781.83

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Worksheet

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A-1440

AAONDraw32 Ver. 4.00 Beta

RL-155-3-0-AB06-000: HGC1-E00-QB1-0B0-JAAC00H-00-0A0000000X
Tag: RTU#1,2Job Name:
Job Number:Winton Medical Center
Job #1Worksheet For:
Worksheet Date:Tobey-Kary Sales Agency
November 18, 2001

Base	Option	Description	List Price	Rep. Price	Cust. Price
R	Series	Roof Top Unit			
L	Generation	Eight Generation			
155	Size	One Hundred and Fifty Five			
3	Voltage	460V/32460Hz	\$0.00	\$0.00	\$0.00
0	Interior Protection	Standard	\$0.00	\$0.00	\$0.00
A	Cooling Style	Drau Thru-R22 Dual Coiled Cores	\$6164.00	\$1721.18	\$1721.18
B	Cooling Configuration	Air Cooled Cond w/ 6R coil High CFM	\$161733.00	\$47795.89	\$47795.89
0	Cooling Coating	Std	\$0.00	\$0.00	\$0.00
6	Cooling Stages	6 Stage	\$1269.00	\$529.20	\$529.20
0	Heating Type	No Heat	\$0.00	\$0.00	\$0.00
0	Heating Ducting	No Heat	\$0.00	\$0.00	\$0.00
0	Refrigerant	No Heat	\$0.00	\$0.00	\$0.00

Feature	Option	Description	List Price	Rep. Price	Cust. Price
H	1A. Outside Air Options	Heat Wheel Small (5) (1-74 inch wheel)*	\$37031.00	\$11048.88	\$11048.88
G	1B. EA Blower Configuration	2 Blowers (Cross or Series/ 2-motors 2-VEF)	\$12467.00	\$3927.10	\$3927.10
C	1C. EA Blower	Blower C (42" Dia 1/2 Horse)	\$3524.00	\$1110.06	\$1110.06
I	1D. EA Motor	25.0 hp (1760 rpm)	\$6824.00	\$1992.08	\$1992.08
E	2. Outside Air Controls	DDC Ktron Control	\$4244.00	\$1537.09	\$1537.09
O	3. Inlet/Exh Location	Bottom Discharge	\$0.00	\$0.00	\$0.00
O	4. Return Location	Bottom Return	\$0.00	\$0.00	\$0.00
Q	5A. SA Blower Configuration	4 Blowers w/ (Cross or Series) w/4-VEF's	\$86115.00	\$2911.22	\$2911.22
B	5B. SA Blower	Blower B (30" Diameter)	\$7904.00	\$2469.76	\$2469.76
I	5C. SA Motor	25.0 hp (1760 rpm)	\$12645.00	\$3984.12	\$3984.12
O	6A. Pre-Filter	X Pleated	\$0.00	\$0.00	\$0.00
B	6B. Floor Filter	12" Cartridge 85% Filt-Filter Box B	\$16730.00	\$4954.95	\$4954.95
O	6C. Filter Options	Std	\$0.00	\$0.00	\$0.00
J	7. Refrigeration Controls	5 MTDR On & Off + 20 STDR + 115V Outlet Factory	\$1403.00	\$442.57	\$442.57
A	8. Refrigeration Options	Hot Gas Expansion Valve (HGV)	\$4870.00	\$1878.55	\$1878.55
A	9. Refrigeration Accessories	Sight Glass	\$1700.00	\$535.50	\$535.50
C	10. Power Options	600 Amps Power Switch	\$8770.00	\$3117.55	\$3117.55
O	11. Safety Options	Std	\$0.00	\$0.00	\$0.00
O	12. Chassis	Std	\$0.00	\$0.00	\$0.00
H	13. Special Controls	Field Installed DDC Controls by Others	\$0.00	\$0.00	\$0.00
O	14A. Pre-Heat Configuration	Std (No Preheat)	\$0.00	\$0.00	\$0.00
O	14B. Pre-Heat Sizing	Std (No Preheat)	\$0.00	\$0.00	\$0.00
O	15. Option Box	Std	\$0.00	\$0.00	\$0.00
A	16. Cabinet Options	Stainless Steel Cabinet Panels	\$4690.00	\$1445.85	\$1445.85
O	17. Cabinet Options	Std	\$0.00	\$0.00	\$0.00
O	18. Compressor Code	Std	\$0.00	\$0.00	\$0.00
O	19. Cool Options	Std ETL USA Listing	\$0.00	\$0.00	\$0.00
O	20. Unit Splits	Std (One Piece 1/2 in)	\$0.00	\$0.00	\$0.00
O	21. Pump & Water Condenser	Std (No Pump or Water Condenser)	\$0.00	\$0.00	\$0.00
O	22. Blank	Std	\$0.00	\$0.00	\$0.00
B	23. Type	Std (Includes 'Gray Paint')	\$0.00	\$0.00	\$0.00
		Subtotal	\$301,341.00	\$94,820.81	\$94,820.81
		Quantity	2		
		Total	\$602,682.00	\$189,741.63	\$189,741.63

AAON, Inc.**A-0440****Submittal**

2425 South Yukon Ave - Tulsa, Oklahoma 74107-2722 - Ph. (918) 625-1266 Fax: (918) 625-6004

AAONSubmittal Ver. 4.00 Rev. 1

RL-155-3-0-AB06-000: HGC1-E00-QB1-0B0-JAAC00H-00-0A000000B
Tag: RTU# 1,2

Job Name:
Job Number:Winton Medical Center
Job #1Submitted For:
Submission Date:Tobey-King Sales Agency
November 19, 2001

Base Option	Description
R Series	Roof Top Unit
L Generation	Eight Generation
155 Size	One Hundred and Fifty Five
3 Volts	460V/32/60Hz
0 Interior Protection	Standard
A Cooling Style	Draw Thru-RTE Dual Circulated Compressor
B Cooling Configuration	Air Cooled Cond w/ 6R coil High CFM
0 Cooling Coating	Std
6 Cooling Stages	6 Stage
0 Heating Type	No Heat
0 Heating Ductwork	No Heat
0 Heating Stages	No Heat

Feature Option	Description
H 1A. Outside Air Options	Hot Wheel Small (5) (1.74 inch wheel)
G 1B. RA Blower Configuration	3 Blowers (From coil mid) w/ 2-motors 2-VFD
C 1C. RA Blower	Blower C (4" Dia 12 Blade)
1 1D. RA Motor	15.0 hp (1760 rpm)
E 2. Outside Air Controls	DDC Econ Control
0 3. Discharge Location	Bottom Discharge
0 4. Return Location	Bottom Return
Q 5A. RA Blower Configuration	4 Blowers w/ (From coil mid) w/4-VFD's
B 5B. RA Blower	Blower B (30" Diameter)
1 6C. RA Motor	25.0 hp (1760 rpm)
0 6A. Pre-Filter	2" Pleated
B 6B. Final Filter	12" Cartridge 45% EK-Filter Box B
0 6C. Filter Options	Std
J 7. Refrigeration Controls	5 MTDR On & Off + 20 STDR + 115V Outlet Factory Wired
A 8. Refrigeration Options	Hot Gas Bypass Load Stage (HGB)
A 9. Refrigeration Accessories	Sight Glass
C 10. Power Options	600 Amps Power Switch
0 11. Safety Options	Std
0 12. Controls	Std
H 13. Special Controls	Field Installed DDC Controls by Contractor
0 14A. Pre-Heat Configuration	Std (No Preheat)
0 14B. Pre-Heat Sizing	Std (No Preheat)
0 15. Option Boxes	Std
A 16. Cabinet Options	Stainless Steel Drain Pans
0 17. Cabinet Options	Std
0 18. Customer Code	Std
0 19. Code Options	Std ETL USA Listing
0 20. Unit Spots	Std (One Piece Unit)
0 21. Even & Water Condenser	Std (No Even or Water Condenser)
0 22. Blank	Std
B 28. Type	Std (Includes 'Glow Paint')

000/006

TUBBY-KING SALES & SERV. + AAON

11/17/01 10:32 AM 412 151 6519

CL 666

AAON, Inc.

A-0440

Unit Rating

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AAONUnit Ver. 4.00 Beta

RL-155-3-0-AB06-000:HGC1-E00-QB1-0B0-JAAC00H-00-0A000000B
Tag: RTU#1,2

Job Information

Job Name: Weirton Medical Center
 Job Number: Job #1
 Site Altitude: 0 ft

Unit Information

Unit Size: One forty five tons - D
 Cabinet Size: 31356 / 81184 lbs.
 Approx. Op./Ship Weights: 32000 / 2.5 in. w.g.
 Supply CFM/ESP: 62000 / 1.30 / 2.75 in. w.g.
 Exhaust CFM/ESP/TSP: 41000 / 1.30 / 2.75 in. w.g.
 Outside CFM: 11500
 Ambient Temperature: 92 °F DB / 75 °F WB

Static Pressure

External: 2.50 in. w.g.
 Evaporator: 0.94 in. w.g.
 Filters Clean: 0.68 in. w.g.
 Dirt Allowance: 0.35 in. w.g.

Economizer: 0.21 in. w.g.
 Heating: 0.00 in. w.g.
 Cabinet: 0.59 in. w.g.
 Heatwheel: 1.30 in. w.g.
 Total: 6.57 in. w.g.

Cooling Section

	Gross	Net
Total Capacity:	1585.76	1363.63 MBH
Sensible Capacity:	1203.04	992.00 MBH
Latent Capacity:	376.73 MBH	
HW Total Cooling Capacity:	6.87 °F MBH	
Mixed Air Temp:	75.00 °F DB	64.00 °F WB
Entering Air Temp:	75.00 °F DB	64.00 °F WB
Lv Air Temp (Cell):	53.71 °F DB	53.64 °F WB
Lv Air Temp (Unit)	57.54 °F DB	55.20 °F WB
Supply Air Fan:	DT - 4 x 300 @ 19.98 BHP Ea.	
SA Fan RPM / Width:	1620 / 99%	
Exhaust Air Fan:	2 x MW4312-35 @ 19.19 BHP Ea.	
EA Fan RPM:	1500	
Evaporator Coils:	86.8 ft ² / 6 Rows / 12 FPI	
Evaporator Face Velocity:	599.0 fpm	
Energy Recovery Wheel:	1 x ERG-7490	

Heating Section

PreHeat Type: Std (No Preheat)
 Heating Type: No Heat

EER - ARI Listing Information

No ARI Rating Program Exists for Units Over 250 MBH
 All AAON Units Are Tested in Accordance With ARI Standards

EER @ ARI Conditions:	9.2	EER Compressor Only @ ARI Conditions:	12.0
Application EER @ Op. Conditions:	5.8	Condensing Unit EER @ Op. Conditions:	11.1

Electrical Data

Rating:	460/3/60	Minimum Circuit Amp:	492
Unit FLA:	481	Maximum Overcurrent:	500

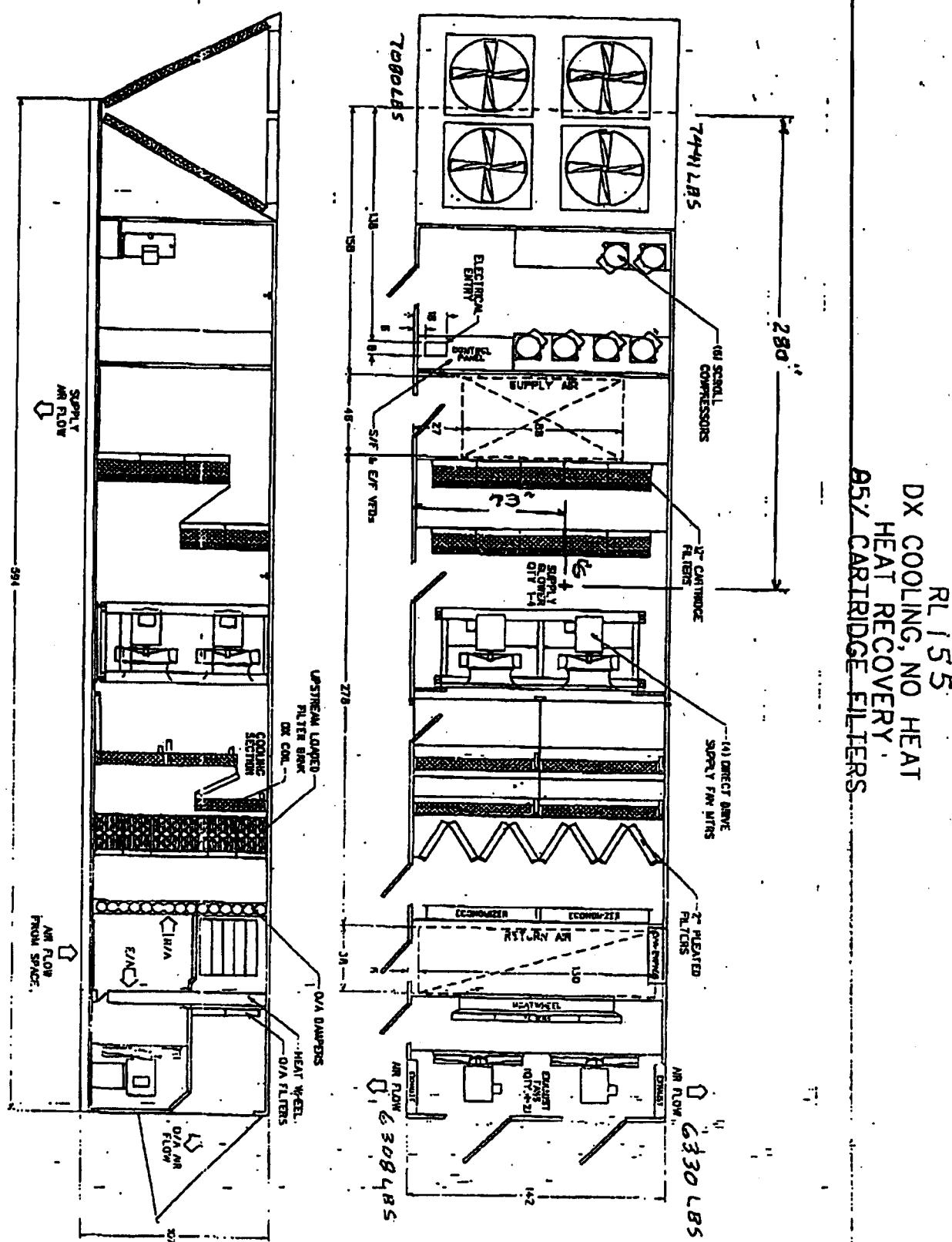
Motors

	Qty	HP	VAC	Phase	RPM	FLA	RLA
Compressor 1:	2	460	3				26.98
Compressor 2:	4	460	3				48.15
Condenser Fan:	5	5.00	460	3	1170	7.3	
Supply Fan:	4	26.00	460	3	1760	34.0	
Exhaust Fan:	2	25.00	460	3	1760	34.0	
Heatwheel:	1	0.25	250	1	1760	0.0	

Cabinet Sound Power Levels*

Octave Bands:	63	126	280	600	1000	2000	4000	8000
Discharge LW(dB):	100	99	99	101	99	97	92	85
Return LW(dB):	98	91	85	77	87	87	72	74

*Sound power levels are given for informational purposes only. The sound levels are not guaranteed.



4-0440

CL 668



REDEMPTION COUPON

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SPECIAL MULTIPLIER DISCOUNT

**FOR
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Issued To:

Rich Nalevanko

**Tobey-Karg Sales Agency
Pittsburgh, PA**

Job Name:

WEIRTON MEDICAL

Signature:

Jeff A. Hall

Date: 11/16/01

Coupon Expires: December 31, 2002

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AAON, Inc.

Sales

SOI Tracking Form

MUST BE COMPLETE BY: 11/19/01

SOLD TO: YONEY/KARG JOB NAME:

AAON CONTACT: REQUESTED SHIP DATE: 3/11/02 (4)

DSO NUMBER: PROJECTED SHIP DATE: 11/19/01

COMMENTS:

Engineering

MUST BE COMPLETE BY:

SOI REQUIRED ?: YES NO
 AEN / ECN No.: N/A
 BILL WORK REQ. ON STD. BILLS ?: YES NO
 ALL UNITS CONFIGURED ?: YES NO

WIRING DIAGRAMS COMPLETE ?: YES NO
 SPECIAL NAMEPLATE INFORMATION: YES NO
 SPECIAL ORDER PARTS REQ. ?: YES NO
 AGENCY APPROVED ?: YES NO
 IF NO AGENCY APPROVAL, EXPLAIN:

COMMENTS:

PREPARED BY: DATE GIVEN TO PROD. / INV. CNTL:

Production / Inventory Control

MUST BE COMPLETE BY:

SPECIAL PARTS ON ORDER ?: YES NO
 SPECIAL BILL WORK COMPLETE ?: YES NO
 ALL WORK ORDERS ENTERED ?: YES NO
 PREPARED BY:

SOI INFO. CORRECT TO ENTER ?: YES NO
 COST ROLL-UP COMPLETE ?: YES NO
 DATE GIVEN TO MANUFACTURING?:
 DATE GIVEN TO SALES:

Manufacturing

MUST BE COMPLETE BY:

SOI COMPLETE ?: YES NO
 NAMEPLATE INFO COMPLETE ?: YES NO
 COMMENTS:

WIRING DIAGRAM COMPLETE ?: YES NO
 SPECIAL INSTRUCTIONS COMPLETE ?: YES NO

PREPARED BY: DATE GIVEN TO PROD. / INV. CNTL:

SPECIAL COMMENTS:

7-11-95

CL 670

AAON, Inc.

Memorandum

via Datafax

To: Creed Hess
Tobey Karg

From: Jim Parro

JP

Date: November 19, 2001

Subject: RL Selection for Weirton Medical Center

cc: B. Pohl
D. Schwartz
M. Roark

R. Schoonover
S. Hammoud

B. Smith
D. Knebel

Confirming our telephone conversation today, I review the following.

We have received and will be entering the subject order for 2 of the RL-155s. As I mentioned to you, we will be adding the net freight amount to the order of \$4608.

Program Error to be Corrected

While you were using the new program, you were able to key the state name directly as "W. Va." rather than using the dropdown box to point and click on "WV".

This disabled the automatic calculation of the freight amount and your printout that was sent to us indicated Zero \$ for freight.

ch111901.doc

CL 671

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Woods Practical Guide to Fan Engineering

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CO-EDITORS

W. C. Osborne, B.Sc. Eng. (Lond.)
C. G. Turner, M.A.A.

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Sixth impression October 1957

SECOND EDITION
June 1960
Second impression March 1961
Third impression August 1964

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SECTION 15**Fans**

THE DEFINITION of a fan is a machine which propels air continuously by aerodynamic action. Piston-type compressors and positive displacement machines in general are not classed as fans. There are three basic types of fans: centrifugal, propeller, and axial flow. The last two are sometimes regarded as a single group, but the differences in their design and characteristics are such that separate classification is warranted.

Desk and ceiling fans are actually of a propeller type, but are not generally included in that category. They do not come into the field of fan engineering proper and are not dealt with in this publication.

CENTRIFUGAL FANS

The centrifugal fan comprises an impeller which rotates in a casing shaped like a scroll as illustrated in fig. 15-1. The impeller has a number of blades or plates around its periphery, similar to a water wheel or the paddle wheel of some shallow draught river steamers. The casing has an inlet on the axis of the wheel and an outlet at right-angles to it as shown in fig. 15-2.

When the impeller rotates the blades at its periphery throw off air centrifugally in a direction following the rotation. The air thrown off into the scroll is forced out of the outlet as more and more leaves the blades. At the same time air is sucked into the inlet to replace that which is discharged. The air enters axially, turns at right-angles through the blades, and is discharged radially. The purpose of the scroll is to convert the high velocity pressure at the blade tips into static pressure.

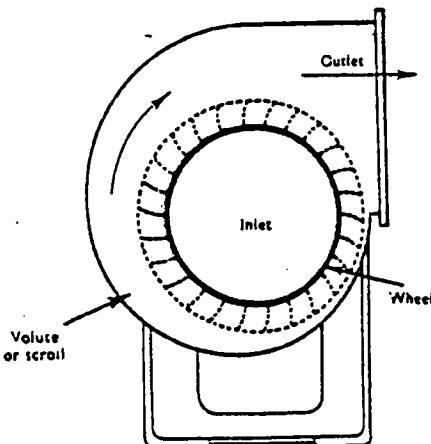


Fig. 15-1. General arrangement of centrifugal fan.

FAN PERFORMANCE

Fans are selected to give a certain quantity of air against a certain pressure and their performance must be defined largely by these two factors. Although designed for optimum performance at a given duty, a fan is capable of working quite reasonably over a range of pressures and volumes, and its performance is more completely defined by a table, or graph of pressure and volume flow of air. This is known as the "characteristic" of the fan. Fig. 15-24 shows a typical pressure-volume characteristic of a 24in. diameter Aerofoil fan with a blade angle of 24° running

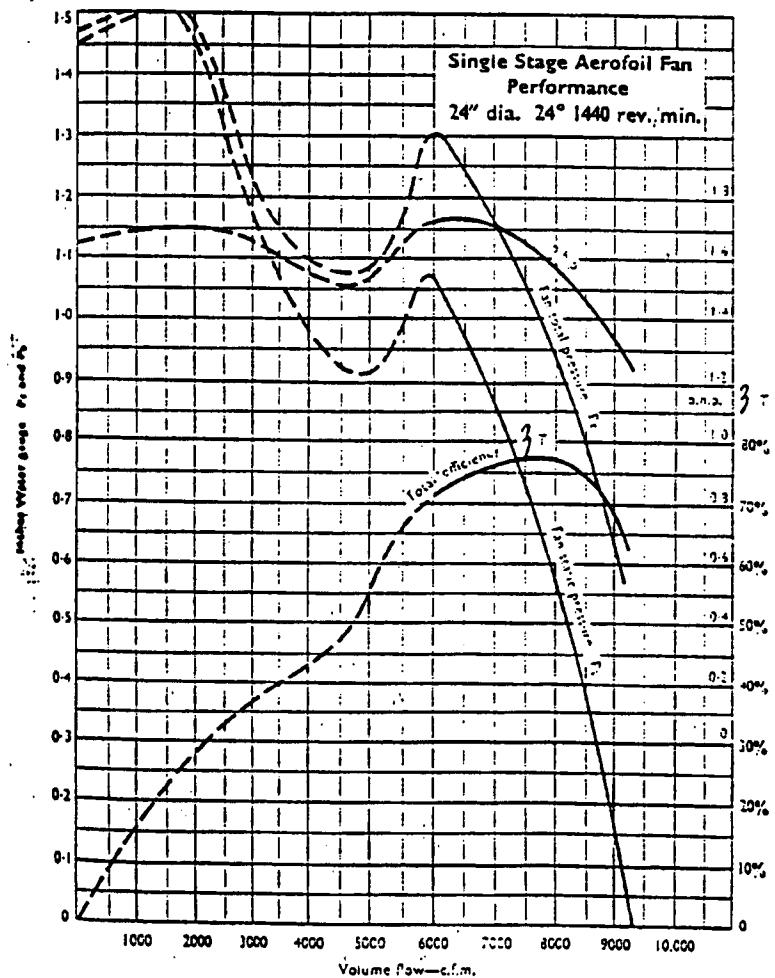


Fig. 15-24. Pressure-volume characteristic of a single-stage axial flow fan

at 1440 rev/min, with additional curves of b.h.p. and fan total efficiency to complete the information. The efficiency curve shown is based on fan total pressure as this is a measure of the total work done on the air.

The relationship between volume, pressure, power and efficiency may conveniently be stated as below, using the following symbols:

Q	... Volume flow of air in unit time—c.f.m.
P_T	... Fan total pressure—in. w.g.
P_S	... Fan static pressure—in. w.g.
b.h.p.	... horse-power absorbed by fan.
η_S	... Fan static efficiency.
η_T	... Fan total efficiency.

b.h.p. absorbed by fan

$$= \frac{\text{Volume flow of air, } Q \text{ c.f.m.} \times \text{fan total pressure, } P_T \text{ in. w.g.}}{6350 \times \text{fan total efficiency}}$$

$$= \frac{\text{Volume flow of air, } Q \text{ c.f.m.} \times \text{fan static pressure, } P_S \text{ in. w.g.}}{6350 \times \text{fan static efficiency}}$$

from which may be derived the relationship

$$\frac{\text{fan static efficiency } \eta_S}{\text{fan total efficiency } \eta_T} = \frac{\text{fan static pressure } P_S}{\text{fan total pressure } P_T}$$

for the same volume flow of air.

Fan laws

Fans are usually made in ranges of size and speed and if, in a given range, each one is identical in all other respects than size to the others, the fans are said to be "geometrically similar". Certain laws govern the relative performance of these fans when working at the same point on the pressure-volume characteristic and may be stated briefly as follows:

With constant impeller size.

1. Volume flow varies directly as the speed of rotation.
2. Pressure developed varies as (speed of rotation)².
3. b.h.p. absorbed varies as (speed of rotation)³.

With constant speed of rotation.

4. Volume flow varies as (impeller size)³.
5. Pressure developed varies as (impeller size)².
6. b.h.p. absorbed varies as (impeller size)³.

Consequently with varying speed of rotation and impeller size.

7. Volume flow varies as (speed of rotation) \times (impeller size)³.

of air density ρ . The constant now is known as K_s or K_T according to whether comparison is being made on static or total pressure.

$$\therefore \text{Static fan pressure } P_s = K_s \times \left(\frac{n}{1000}\right)^2 \times (d \text{ in ft.})^2 \times \rho_s$$

$$\text{Total fan pressure } P_T = K_T \times \left(\frac{n}{1000}\right)^2 \times (d \text{ in ft.})^2 \times \rho_s$$

$$\text{Similarly b.h.p.} = K_p \times \left(\frac{n}{1000}\right)^3 \times (d \text{ in ft.})^3 \times \rho_s$$

K_Q , K_s and K_T , and K_p represent the volume flow, pressure and b.h.p. of a one ft. dia. fan running at 1000 rev/min with air at standard density. From the equations on page 138 it follows that:

$$K_p = \frac{K_Q \times K_s}{6350 \times \eta_s} \text{ where } \eta_s = \text{fan static efficiency}$$

$$= \frac{K_Q \times K_T}{6350 \times \eta_T} \text{ where } \eta_T = \text{fan total efficiency}$$

If fan performance is now plotted in terms of K_Q , K_s , K_T , and K_p instead of volume flow, fan static, and total pressures, and b.h.p. a basis of comparison between fans of different series is readily available, the shape of the "standard" characteristic being in every way identical with that of any fan of the same series. Figs. 15-25 and 15-26 show the characteristics of a type J Aerofoil fan, one of 24in. diameter running at 1440 rev/min and the other in terms of coefficients K_Q and K_s .

REVERSIBILITY OF FANS

In many ventilating and air circulating systems it is desirable at some time to reverse the direction of air flow. Sometimes this is done as an emergency measure, and in some cases to prevent stagnation of air in such places as ships' cargo spaces and refrigerated spaces.

If centrifugal fans are employed reversal will entail a rather complicated system of ductwork, which provides by means of doors an alternative path for the air. Reversal of air flow from centrifugal fans is impossible by any other means, as they are essentially non-reversible.

Propeller and axial flow fans are, however, essentially reversible fans, though, depending upon the individual design, some are more effective than others. Reversal of air flow is simply achieved by reversing the direction of rotation and in the case of electrically driven fans, by means of a switch. This method may be applied to non-guide-vane single-stage fans and to contra-rotating fans, but fans with guide vanes are generally unsuited to this method of reversal.

With the usual types of propeller and axial flow fans, a reduced volume is delivered when the impeller runs in the reverse direction, and is generally from 70% to 75% for propeller and single stage fans and 65% to 70% of the forward volume for contra-rotating fans when operating on the same system of ductwork.

Where equal volume is required in both directions, special fans such as truly reversible Aerofoil fans can be constructed. These have the impeller blades assembled with aerofoil sections set alternately in opposite directions. Thus, whichever way the impeller rotates the conditions of running are the same, and therefore the same volume flow of air results.

It is obvious that some reduction in performance compared with the air delivery given by the standard design is inevitable, but this reduction is not great. For instance, by comparison with a standard Aerofoil fan of the same size and speed, a Truly Reversible Aerofoil fan will deliver about 85% of the volume against about 70% of the pressure. The total efficiency of such a fan is quite high, being 60% to 65%, compared with 70% to 78% of the comparable standard fan.

OPERATION OF FANS IN PARALLEL

Identical fans may be operated quite satisfactorily in parallel when two such fans will deliver twice the volume of air at the same pressure as a single unit. Non-identical fans, too, may be operated in parallel, but care must be taken to select a good working position on the combined characteristic and even then maximum efficiency is unlikely to be achieved at the same time by each fan.

If, as in the case of an axial flow fan of high blade pitch angle, stalling characteristics are exhibited at high pressure, the combined unit will also

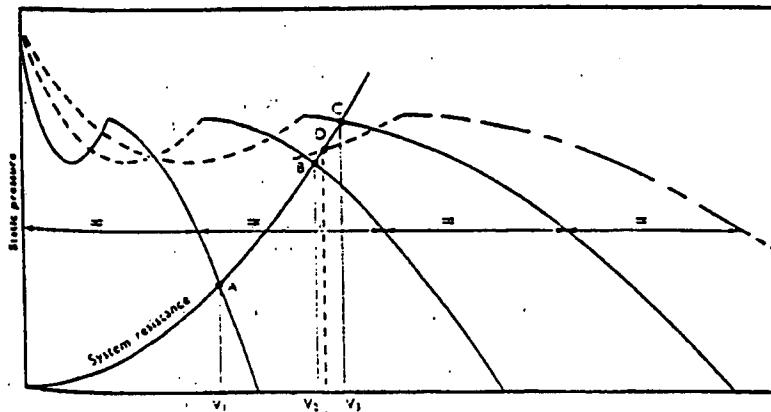


Fig. 15-27. Volume and pressure characteristics of fans in parallel

exhibit these characteristics. Consequently, care must be taken in selection of fans for parallel operation to avoid this possibility. The danger is probably greatest when it is desired to add another fan to the system, in which case, as is illustrated on page 147 by point D, the point of working on the combined characteristic may easily be changed from a perfectly satisfactory one to a very unsatisfactory one.

Two fans operating on the same system, it should be noted, do not give twice as much air as one of them would give when working alone on the system. As the resistance of the system usually increases as the (volume flow of air)², the latter settles down at some value which is less than twice the volume given by one fan. The increase in volume per extra fan decreases as the number of fans working in parallel is increased.

A form of volume control is feasible by switching off one or more units, but generally it will be necessary to provide anti-backdraught devices to prevent short circuiting of the air back through the fans not in use.

Fans are usually operated in parallel when lack of space forbids erection of a single large fan. Sometimes, too, a number of small fans may be installed at a lower capital expenditure than a single unit capable of the combined duty. Moreover, the risk of complete shut down is minimised as individual fans may be taken out of service for maintenance without closing down the system, provided shutters are available for blanking off the apertures of the shut-down fans.

VOLUME REGULATION OF FANS

In many fan systems some control of the volume flow of air is desirable. This may be done by any one of many methods, though some methods are much more desirable than others. From the point of view of power consumption, the ideal method is to vary the speed of the fan, although in practice even this cannot be achieved without some loss of power. Fig. 15-29 gives an idea of the relative merits of the following types of volume control compared on a power consumption basis with the ideal method of speed regulation.

Damper control

This is a method very widely used as it is provided by some simple method of throttling the air flow in the system. In other words, the system resistance is varied by

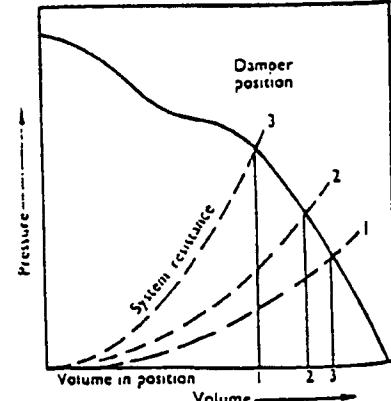


Fig. 15-28. Effect of damper control

frequency distribution of the main fan will be lower than that of a fan designed to supply a single 9in. \times 9in. duct. The sound power level at each outlet is therefore:

$$79 - 9 - 1 - 2 = 67 \text{ dBp.}$$

Sound absorbers

Proprietary absorbers or silencers usually sub-divide the area available for air flow into several passages each lined with perforated sheet backed by rock-wool, glass fibre or some other sound-absorbing material. The attenuation in decibels should be quoted, preferably in octave bands of frequency so that the degree of match with the frequency distribution of the fan may be gauged. Resistance to air flow must also be considered since it is clearly unsatisfactory to absorb so much pressure (inches w.g.) that the fan speed has to be put up, thereby generating more sound and incurring additional power consumption.

Absorbers may be built to suit an installation by inserting into the duct splitters having perforated walls and packed with absorbent material. To be effective on both faces the splitter needs to be twice the thickness of the equivalent duct lining. The benefit obtained from the use of splitters lies primarily in reduction of length since the same amount of absorbent material used as a simple lining may be equally effective if the necessary length is available.

Fig. 18-9 shows some examples of lined ducts which will pass equal volumes of air for the same pressure drop, and will also provide equal noise reduction according to the formula usually employed. This may be written:

$$dB = 4.2 a^{1.4} L (4A/P)$$

L = Length in direction of air flow, in.

P = Perimeter of cross-section, in.

A = Area of cross-section, sq. in.

4A/P = Diameter of equivalent circular section,

= Length of side of square section,

= Twice width of very elongated rectangular section.

a is not simply the normal absorption coefficient of the particular lining employed. It is also a complex function of the shape and size of the duct and the sound frequency. Fig. 18-10 illustrates some effective values of $a^{1.4}$ which have been found experimentally.

SECTION 20

Backdraught prevention

THE EFFECT of opposing winds must be considered when fans exhaust to atmosphere through a hole in a wall. Wind blowing against a fan outlet may restrict the air output and also cause objectionable draughts through the fan aperture. For both these reasons it is advisable to protect the fan with a shutter or cowl.

There are two types of automatic shutter for preventing backdraught when the fan is switched off—louvre and butterfly. The louvre shutter is the cheaper of the two, but its suitability is somewhat limited. This type comprises metal vanes pivoted in a steel ring, as illustrated in fig. 20-1. The vanes are opened by the fan draught and they close by gravity when the fan draught stops. To prevent undue restriction of air output the vanes must open to a minimum angle of 60 degrees. This normally requires a velocity of 1000ft. to 1200ft. per minute. If the discharge velocity is less the shutter vanes will not open sufficiently and the fan output will be restricted. Louvre shutters are therefore not suitable for fans with low outlet velocities. Nor are they to be recommended for high speed fans. The reason for this is that high velocities through the shutter make the vanes rattle. Due to this they may be objectionably noisy, and in the course of time they may even disintegrate. As a general rule louvre shutters are recommended for fan output velocities between 1200ft. and 1500ft. per minute.

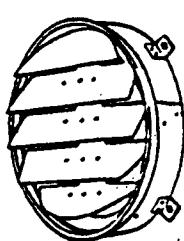


Fig. 20-1.
Louvre shutter

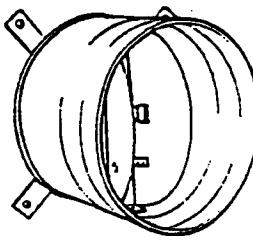


Fig. 20-2. Butterfly shutter

The butterfly shutter does not present the same limitations. This comprises a cylindrical barrel in which two semi-circular flaps are pivoted at an angle. The flaps open in the fan draught and close by gravity when the fan

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Selecting Fans—Determining Airflow for Crop Drying, Cooling, Storage

COLLEGE OF AGRICULTURAL, FOOD, AND ENVIRONMENTAL SCIENCE

William F. Wilcke, Extension Engineer

R. Vance Morey, Professor and Head, Biosystems and Agricultural Engineering Department

Using fans to force air having the proper temperature and relative humidity through a crop is a valuable technique for maintaining quality after harvest. The air helps maintain the moisture, temperature, and oxygen content of a crop at levels that prevent growth of harmful bacteria and fungi and excessive shrinkage.

This fact sheet provides information that will help you select new fans for crop drying, cooling, or storage facilities, or help you determine airflow delivered by existing fans. Grains and oilseeds are the primary crops discussed, but hay, potatoes, and other types of produce are also mentioned.

Airflow Requirements

Total airflow provided by a fan is usually expressed as cubic feet of air per minute (cfm). Recommendations for drying or aerating a particular crop are given as airflow per unit of crop being served by the fan. For example, cfm per bushel (cfm/bu) is used for drying or aerating grains and oilseeds. Typical airflow recommendations are listed in Table 1. Select fans that deliver airflow within the ranges given in the table: greater airflows require larger fans and lead to greater costs, while lower airflows could result in unacceptable crop quality.

Airflow Resistance

Crops

When air is forced through a bulk crop, it must travel through narrow paths between individual particles. For packaged crops, air must travel through or between individual containers. Friction along air paths creates resistance to airflow. Fans must develop enough pressure to overcome this resistance and move air through the crop.

Airflow resistance and fan pressure are usually expressed in inches of water column (in. water, or in. H_2O). This term comes from gages called u-tube manometers that are sometimes used to measure pressure (Figure 1). You can make a u-tube manometer by fastening a clear plastic tube and a ruler to a board. Then pour some water, or water plus a small amount of antifreeze, into the tube. Since manometers are used to measure pressure relative to atmospheric pressure, leave one end of the tube open to the atmosphere. Attach the other end to the duct or plenum where you

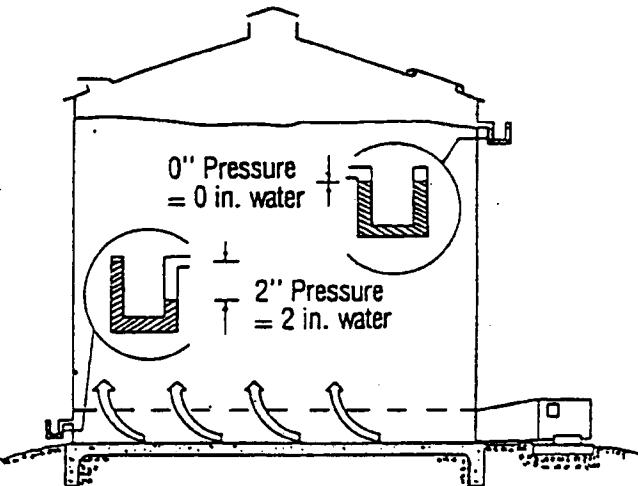


Figure 1. Using a u-tube manometer to measure pressure in a grain bin.

want to measure pressure. When a fan generates pressure, it forces water in the tube to move in the direction of lower pressure. The height difference of the water levels on the two sides of the tube, measured in inches, is the fan static pressure, in. water. In negative pressure or suction systems, pressure between the crop and the fan is less than atmospheric pressure and water in the manometer tube moves toward the fan. In positive pressure systems, water moves away from the fan. You can buy dial-type pressure gauges that operate on a different principle but that are calibrated to give readings in. water.

The airflow resistance of a crop and the fan pressure required to overcome it depend on how fast the air is moving and how long and narrow the paths are. For grains and oilseeds, these factors are a function of the particular crop (size and shape of seeds), crop depth, and airflow rate (cfm/bu) you're trying to provide.

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As you can see from Tables 2 through 6, at a given airflow rate, crop depth has a large effect on static pressure. Static pressure, in turn, greatly affects fan power requirements. Short, large diameter bins are recommended for natural-air grain drying because static pressure and required fan size are smaller than they would be in tall, narrow bins. Even though short bins cost more to install than tall ones that have the same grain capacity, total drying costs are less because smaller fans use less electricity.

Airflow resistance of hay, potatoes, and other produce also depends on crop depth or thickness of the layer to be ventilated and airflow rate. For packaged produce, the type of container and the way containers are stacked can also make a difference. But in most cases, airflow resistance of these crops seldom requires fan pressure greater than about 1 in. water. If better information is lacking, use 1 in. as a static pressure estimate for these crops.

Floors and Ducts

The full perforated floors used in grain bins generally have negligible resistance to airflow. Airflow resistance of bin floors isn't significant unless open area is less than about 7%; most commercially available floors have more than 10% open area.

Air supply ducts, tunnels, and perforated air distribution ducts offer greater resistance to airflow than do full perforated floors. This resistance can be quite large if ducts are too small or too long. Use ducts that are large enough that air velocity is less than about 1500 feet per minute. (Divide duct airflow in cfm by duct cross sectional area in square feet to get velocity.) Also, try to keep duct length less than 100 ft. Unless you have better information, use 0.5 in. water as an estimate of airflow resistance for duct systems. Be aware that corrugated plastic ducts designed for air distribution have only 1 to 3% open area, and ordinary plastic tile designed for field drainage has less than 1% open area. Because plastic ducts have so little area for air exit, their airflow resistance can exceed 0.5 in. water.

Air inlet and exhaust openings

When outdoor air is used to ventilate a bin or building, you need to provide adequately-sized openings for air to move in and out of the structure. If openings are too small, they restrict airflow and increase fan pressure requirements. Provide at least one square foot of inlet area per 1000 cfm and an equal exhaust area, and make sure these vents or doors are open anytime the fan is operating.

Fan Performance

Because of the way fan impellers (blades or rotors) are designed, the amount of air they can move decreases as the pressure they are working against increases. The airflow vs. pressure information for a particular fan is

called the fan performance data. Performance depends on the size, shape, and speed of the impeller, and the size of the motor driving it. Performance differs widely among brands and models, even for fans with the same size motor.

Access to fan performance data is essential for selecting fans and for determining airflow provided by existing fans. Most manufacturers sell fans that have been tested using procedures specified by the Air Movement and Control Association International, Inc. (AMCA). The manufacturers can provide you with performance data in the form of tables or graphs. Avoid fans for which AMCA data is not available. Table 7 is an example of the type of data you need. Figure 2 is a graphical presentation of the data for two fans from Table 7 that have the same size motor. Note how much performance of the two fans differs.

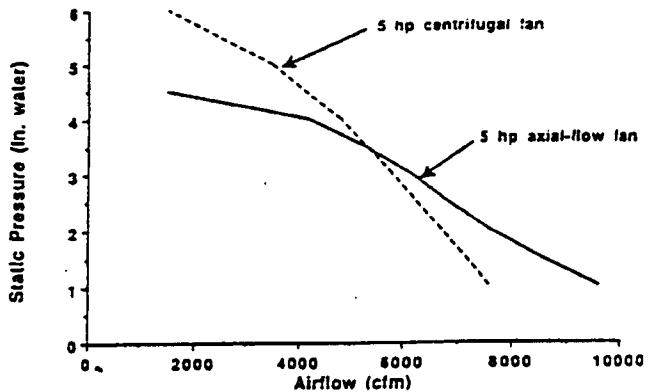


Figure 2. Fan performance data for MES Fans #7 and #10 from Table 7.

Fan Types

Most fans can be categorized as either axial-flow or centrifugal (see Figure 3). Axial-flow fans are sometimes called propeller fans, although that's really just one type of axial-flow fan. Air moves in a straight line through axial-flow fans parallel to the axis or impeller shaft. The impeller has a number of blades attached to a central hub.

Centrifugal fans are sometimes called blowers or squirrel cage fans. The impeller is a wheel that consists of two rings with a number of blades attached between them. Air enters one or both ends of the impeller parallel to the shaft and exits one side perpendicular to the shaft. The blades can be straight, slanted in the direction of airflow (forward-curved), or slanted opposite the airflow direction (backward-curved or backward-inclined).

AXIAL-FLOW FANS

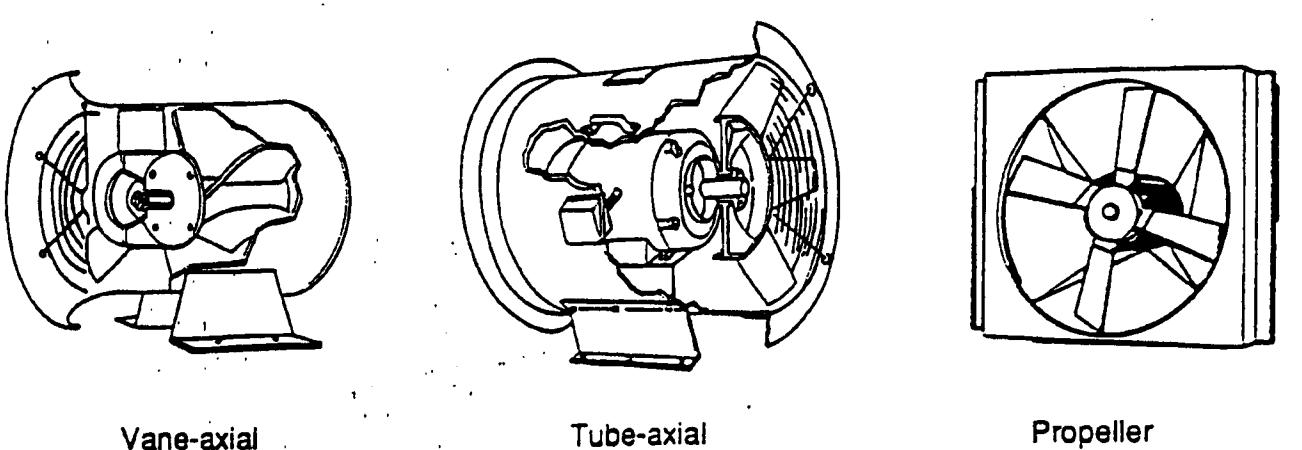


Figure 3. Types of fans used for ventilating crops.

Propeller Fans (panel fans)

These are axial-flow type fans that have from two to about seven long blades attached to a small hub. Fan diameter is usually large relative to the fan's length or thickness. Some propeller fans are called panel fans and are designed for mounting in a wall or plenum divider. Some are belt-driven and some have the impeller hub attached directly to the motor shaft (direct-driven).

Propeller fans normally can't generate more than about 2 in. water pressure.

They are most commonly used for potato ventilation, forced-air produce cooling, hay drying, exhausting air from attics or overhead spaces, or general air circulation. They are seldom used for grain drying or aeration.

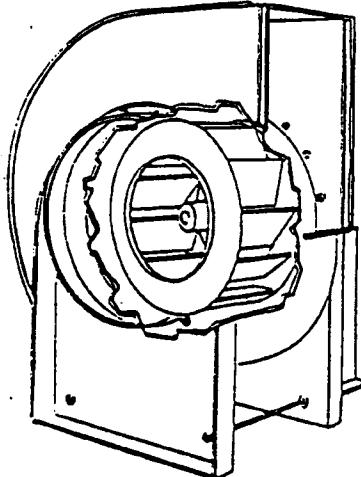
Tube-axial, vane-axial

These axial-flow fans have a barrel-shaped housing and an impeller that has a large hub with a number of short blades attached to it. They are generally direct-driven and the motor is cooled by the airstream. In positive pressure systems, the air stream captures the waste heat given off by the motor. Vane-axial fans have guide vanes inside the fan housing to help reduce air turbulence.

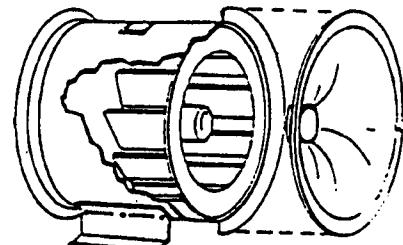
Tube-axial and vane-axial fans are the most common types used for grain drying and aeration. They are relatively inexpensive and fairly efficient when static pressure is less than about 4 in. water. The main disadvantage of these fans is that they are very noisy.

Tube-axial

Propeller



CENTRIFUGAL FANS



Backward-inclined centrifugal

In-line centrifugal

Centrifugal

The centrifugal fans used for crop drying and storage generally have backward-curved or backward-inclined blades. They are expensive, but are also quiet and are usually the most efficient type of fan when static pressure is greater than about 4 in. water. The motor on centrifugal fans is normally outside the air stream; you need to install a special housing around the motor if you want to capture the heat it gives off.

Forced-air heating and ventilating systems often use centrifugal fans that have forward-curved blades. Motors on these fans can be overloaded and burn out when the fans are operated outside certain pressure ranges. This characteristic makes them unsuitable for many crop drying and storage applications.

In-line centrifugal

These fans have axial airflow, but use a centrifugal-type impeller. Price and operating characteristics are between those of backward-inclined centrifugal and tube-axial fans.

Multiple Fans

It is sometimes necessary or desirable to install more than one fan to provide air to a common plenum or supply manifold for a duct system. Fans can be arranged in parallel or series (Figure 4). Reasons for using multiple fans include:

- Total airflow, pressure, or power requirements exceed the capabilities of the largest fan available from your dealer.
- The starting current for a single large fan motor is greater than the electrical system can handle. The maximum starting current is lower if several small fans are started one at a time.
- When multiple fans are installed, you have the option of turning some of the fans off and operating with a lower airflow when conditions allow.
- Air distribution is sometimes more uniform when several small fans are used in place of one large one.

Parallel

Parallel arrangement means fans are installed side-by-side or at several points along a manifold or plenum. The most common applications are where total airflow requirement is large, but pressure is moderate. When fans are installed in parallel, they all face the same pressure. Total airflow is estimated by adding the airflow provided by each fan at the expected pressure.

Series

Series arrangement, where fans are fastened in line or end-to-end, is not used very often. When it is used, it generally involves tube-axial or vane-axial fans in situations where pressure is relatively high, such as in deep grain bins. Series arrangement is seldom used with centrifugal fans and seldom are more than two axial-flow fans connected in series. When fans are arranged in series, each fan handles the same airflow. Total pressure is estimated by adding the pressure developed by each fan at the expected airflow.

Determining Air Flow Provided by Existing Fans

Knowledge of the airflow that a fan is providing allows you to estimate the time it will take to dry or cool a crop. This in turn, helps you determine whether steps need to be taken to prevent unacceptable quality loss before the task is completed.

The first step in determining airflow is to measure static pressure in the duct or plenum between the fan and the crop (Figure 1). Drill a small hole (1/8 in. should be adequate) in the wall of the duct or plenum and press a tube from one side of a pressure gauge or u-tube manometer against the hole. Then, take the pressure reading and use its absolute value (this means assume the reading is positive even if it's a negative pressure system) to determine the airflow. Use the AMCA performance data for that model fan at that

pressure. To get airflow rate (cfm/bu, for example), divide the airflow from the performance table or graph by the amount of crop being served by the fan.

For example, suppose fan #4 from Table 7 is being used to dry 10 tons of hay and the static pressure reading in the duct to which the fan is attached is 1.0 in. water. The fan performance data in Table 7 shows that fan #4 provides 2775 cfm against a pressure of 1 in. Airflow per ton is $2775 \text{ cfm} / 10 \text{ tons} = \text{about } 278 \text{ cfm/ton}$. This value is within the recommended range for hay drying given in Table 1.

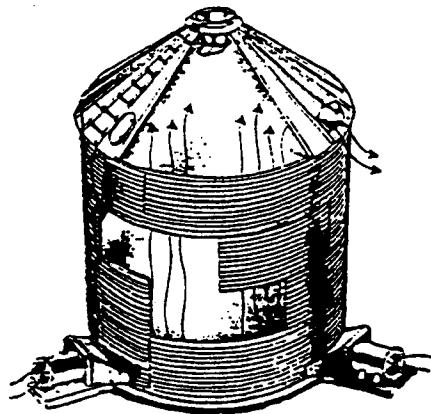
Because airflow resistance and static pressure vary with type of crop, crop depth, amount of fines present, and the way the crop is piled, you need to repeat the above procedure and determine a new airflow anytime conditions change.

Selecting Fans

Calculate total airflow needed

The first step in selecting a fan is to determine the total airflow it must provide. You can use the airflow rates in Table 1 as a guide. Choose an airflow rate, estimate the total quantity of crop to be served by the fan, and then multiply the airflow rate by crop quantity to get total airflow requirement.

PARALLEL



SERIES

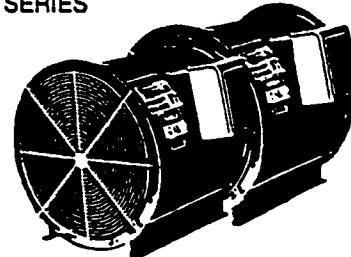


Figure 4. Parallel and series fan arrangement.

For example, if you want to supply 1 cfm/bu to natural-air dry corn in a 27-ft diameter by 16 ft deep bin that has a full perforated floor, calculate airflow as follows:

$$\begin{aligned}\text{Bin capacity} &= (\pi + 4) \times (\text{diameter})^2 \times \text{depth} \times 0.8 \text{ bu/cu ft} \\ &= 0.785 \times 27 \text{ ft} \times 27 \text{ ft} \times 16 \text{ ft} \times 0.8 \text{ bu/cu ft} \\ &= 7325 \text{ bu}\end{aligned}$$

$$\text{Total airflow} = 1 \text{ cfm/bu} \times 7325 \text{ bu} = 7325 \text{ cfm}$$

Estimate static pressure

The next step in selecting a fan is to estimate the pressure the fan will be operating against. For grains and oilseeds, use the desired airflow rate and expected crop depth and read the appropriate pressure value from Tables 2 through 6. Remember to add 0.5 in. to the value from the table if air is distributed through a duct system. For hay, potatoes, or other produce, use 1 in. water as a pressure estimate unless a better number is available.

Continuing our example, Table 3 indicates that the expected pressure for 16 ft of corn and an airflow rate of 1 cfm/bu is 2.4 in. water.

Estimating fan power requirements

Fans are usually described by the horsepower (hp) rating of the motor used to drive the impeller. It's helpful when selecting fans to estimate the power requirement first so you know where to start looking in the manufacturer's catalog.

Fan motor size depends on the total airflow being delivered, the pressure developed, and the impeller's efficiency. Impeller efficiencies generally range from 40% to 65%. If we assume an average value of 60%, we can use the following formula to estimate the fan power requirement.

$$\text{Fan power (hp)} = \text{airflow (cfm)} \times \text{static pressure (in. water)} + 3814$$

In our example,

$$\text{Fan power} = 7325 \text{ cfm} \times 2.4 \text{ in. water} + 3814 = 4.6 \text{ hp.}$$

Selecting the best fan available

Purchase cost and noise during operation can be important factors in selecting a fan, but the most critical factor is whether the fan can provide enough airflow at the expected operating pressure. Start by looking at performance data for a fan having a motor rated just under the power value you calculated. If this fan provides more than enough airflow, look at the next size smaller. If your first pick is too small, try the next size larger.

If we use the list of fans in Table 7 to select a fan for our example problem, we see that fan #7 (a 5.0-hp axial flow fan) comes closest to meeting our needs. Fans #6

and #10 wouldn't provide enough airflow at 2.4 in. water and fans #8 and #11 would provide much more airflow than is needed.

Sometimes fans produced by one manufacturer won't meet your needs and you'll have to look at another manufacturer's fans. Or, if you are having trouble finding a fan that is big enough, you might consider using several smaller fans. (See the section on multiple fans.)

Computerized fan selection

The fan selection procedure that was just described is not too difficult, but there is an easier way to select fans for grain bins.

You can use the FANS or WINFANS (Windows version) computer programs available from the University of Minnesota Biosystems and Agricultural Engineering Department and some county Extension offices. The program is very user friendly and guides you through the fan selection process by asking some simple questions about your grain drying or storage bin. If you have access to the World Wide Web, the program can be downloaded from: www.bae.umn.edu/extens/harvest.html. The program allows you to select fans from a list of over 200 commercially available models and see if the selected models provide the desired airflow.

Summary

Selection of proper fans and determination of actual airflow provided by existing fans are important steps in preserving quality of crops after harvest. Make sure you have fans that provide enough airflow to dry or cool crops before unacceptable quality loss occurs. Contact your local extension office for more information on selecting fans or managing crops after harvest.

Table I. Airflow recommendations for drying, cooling, and storing crops.

Natural-air drying grains & oilseeds	0.75 to 1.5 cfm/bu
Aeration of stored dry grains & oilseeds	0.05 to 0.5 cfm/bu
Hay drying	150 to 500 cfm/ton
Potato ventilation (airflow per hundredweight)	0.5 to 1.5 cfm/cwt
Forced-air produce cooling	1 to 10 cfm/lb

Table 2. Airflow resistance data for barley and oats.

Values in the table have been multiplied by 1.5 to account for fines and packing in the bin. Add 0.5 in. water to the table values if air is distributed through a duct system.

Grain Depth (ft)	Airflow (cfm/bu)								
	0.05	0.1	0.25	0.5	0.75	1.0	1.25	1.5	2.0
Expected static pressure (inches of water)									
2	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1
4	0.1	0.1	0.1	0.1	0.2	0.2	0.3	0.3	0.5
6	0.1	0.1	0.1	0.2	0.4	0.5	0.7	0.8	1.1
8	0.1	0.1	0.2	0.4	0.7	0.9	1.2	1.5	2.1
10	0.1	0.1	0.3	0.7	1.1	1.5	2.0	2.5	3.6
12	0.1	0.2	0.5	1.0	1.6	2.3	3.0	3.7	5.4
14	0.1	0.3	0.7	1.4	2.2	3.2	4.2	5.3	7.8
16	0.2	0.3	0.9	1.9	3.0	4.3	5.7	7.2	10.6
18	0.2	0.4	1.1	2.4	3.9	5.6	7.5	9.5	14.1
20	0.3	0.5	1.4	3.0	4.9	7.1	9.5	12.2	18.1
15	0.4	0.8	2.2	4.9	8.2	11.9	16.1	20.7	31.1
30	0.6	1.2	3.2	7.4	12.4	18.3	24.8	32.1	48.7
40	1.0	2.1	6.0	14.2	24.4	36.2	49.8	*	*
50	1.6	3.4	9.9	23.8	41.4	*	*	*	*

* Static pressure is excessive—greater than 50 in. water.

Table 3. Airflow resistance data for shelled corn.

Values in the table have been multiplied by 1.5 to account for fines and packing in the bin. (If corn is stirred, which tends to decrease airflow resistance, divide table values by 1.5.) Add 0.5 in. water to the table values if air is distributed through a duct system.

Grain Depth (ft)	Airflow (cfm/bu)								
	0.05	0.1	0.25	0.5	0.75	1.0	1.25	1.5	2.0
Expected static pressure (inches of water)									
2	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1
4	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.2	0.2
6	0.1	0.1	0.1	0.1	0.2	0.3	0.3	0.4	0.6
8	0.1	0.1	0.1	0.2	0.3	0.5	0.6	0.8	1.2
10	0.1	0.1	0.2	0.3	0.5	0.8	1.1	1.4	2.0
12	0.1	0.1	0.2	0.5	0.8	1.2	1.6	2.1	3.2
14	0.1	0.1	0.3	0.7	1.2	1.7	2.3	3.0	4.6
16	0.1	0.1	0.4	0.9	1.6	2.4	3.2	4.2	6.4
18	0.1	0.2	0.5	1.2	2.1	3.1	4.3	5.6	8.7
20	0.1	0.2	0.7	1.6	2.7	4.0	5.6	7.3	11.3
25	0.2	0.4	1.1	2.6	4.6	7.0	9.7	12.8	19.9
30	0.3	0.5	1.6	4.1	7.2	11.0	15.3	20.3	31.9
40	0.5	1.0	3.1	8.1	14.6	22.6	31.9	42.5	*
50	0.7	1.6	5.3	14.0	25.6	39.9	*	*	*

* Static pressure is excessive—greater than 50 in. water.

Table 4. Airflow resistance data for soybeans and confectionery sunflowers.

Values in the table have been multiplied by 1.5 to account for fines and packing in the bin. Add 0.5 in. water to the table values if air is distributed through a duct system.

Grain Depth (ft)	Airflow (cfm/bu)								
	0.05	0.1	0.25	0.5	0.75	1.0	1.25	1.5	2.0
	Expected static pressure (inches of water)								
2	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1
4	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.2
6	0.1	0.1	0.1	0.1	0.2	0.2	0.3	0.3	0.5
8	0.1	0.1	0.1	0.2	0.3	0.4	0.5	0.6	0.9
10	0.1	0.1	0.1	0.3	0.4	0.6	0.8	1.0	1.5
12	0.1	0.1	0.2	0.4	0.7	0.9	1.2	1.6	2.3
14	0.1	0.1	0.3	0.6	0.9	1.3	1.7	2.2	3.3
16	0.1	0.1	0.3	0.8	1.2	1.8	2.4	3.0	4.5
18	0.1	0.2	0.4	1.0	1.6	2.3	3.1	4.0	6.0
20	0.1	0.2	0.6	1.2	2.0	3.0	4.0	5.1	7.7
25	0.2	0.3	0.9	2.0	3.4	5.0	6.8	8.8	13.4
30	0.2	0.5	1.3	3.1	5.2	7.7	10.6	13.7	21.0
40	0.4	0.9	2.5	5.9	10.3	15.4	21.4	28.0	43.4
50	0.6	1.4	4.1	10.0	17.6	26.7	37.2	49.1	*

* Static pressure is excessive—greater than 50 in. water.

Table 5. Airflow resistance data for oil-type sunflowers.

Values in the table have been multiplied by 1.5 to account for fines and packing in the bin. Add 0.5 in. water to the table values if air is distributed through a duct system.

Grain Depth (ft)	Airflow (cfm/bu)								
	0.05	0.1	0.25	0.5	0.75	1.0	1.25	1.5	2.0
	Expected static pressure (inches of water)								
2	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1
4	0.1	0.1	0.1	0.1	0.1	0.2	0.2	0.2	0.3
6	0.1	0.1	0.1	0.2	0.3	0.4	0.5	0.6	0.9
8	0.1	0.1	0.1	0.3	0.5	0.7	0.9	1.1	1.7
10	0.1	0.1	0.2	0.5	0.8	1.1	1.5	1.9	2.8
12	0.1	0.1	0.3	0.7	1.2	1.7	2.3	2.9	4.4
14	0.1	0.2	0.5	1.0	1.7	2.4	3.3	4.2	6.4
16	0.1	0.2	0.6	1.4	2.3	3.3	4.5	5.8	8.8
18	0.1	0.3	0.8	1.8	3.0	4.4	6.0	7.8	11.8
20	0.2	0.3	1.0	2.3	3.8	5.6	7.7	10.0	15.3
25	0.3	0.6	1.6	3.7	6.5	9.7	13.3	17.4	26.9
30	0.4	0.8	2.4	5.7	10.0	15.1	20.9	27.5	42.7
40	0.7	1.5	4.5	11.3	20.1	30.7	43.0	*	*
50	1.1	2.4	7.5	19.3	34.8	*	*	*	*

* Static pressure is excessive—greater than 50 in. water.

Table 6. Airflow resistance data for wheat and sorghum.

Values in the table have been multiplied by 1.3 for wheat and 1.5 for sorghum to account for fines and packing in the bin. Add 0.5 in. water to the table values if air is distributed through a duct system.

Grain Depth (ft)	Airflow (cfm/bu)								
	0.05	0.1	0.25	0.5	0.75	1.0	1.25	1.5	2.0
Expected static pressure (inches of water)									
2	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.2
4	0.1	0.1	0.1	0.2	0.3	0.3	0.4	0.5	0.7
6	0.1	0.1	0.2	0.4	0.6	0.8	1.0	1.2	1.7
8	0.1	0.1	0.3	0.7	1.1	1.5	1.9	2.3	3.2
10	0.1	0.2	0.5	1.1	1.7	2.3	3.0	3.7	5.3
12	0.1	0.3	0.8	1.6	2.5	3.4	4.5	5.6	7.9
14	0.2	0.4	1.0	2.2	3.4	4.8	6.3	7.8	11.3
16	0.3	0.5	1.4	2.9	4.6	6.4	8.4	10.6	15.3
18	0.3	0.7	1.7	3.7	5.9	8.3	11.0	13.8	20.0
20	0.4	0.8	2.2	4.7	7.5	10.5	13.9	17.6	25.6
25	0.6	1.3	3.4	7.5	12.2	17.4	23.1	29.4	43.3
30	0.9	1.9	5.1	11.2	18.3	26.3	35.3	45.0	*
40	1.7	3.4	9.3	21.1	35.1	*	*	*	*
50	2.6	5.4	15.0	34.8	*	*	*	*	*

* Static pressure is excessive—greater than 50 in. water.

Table 7. Example of fan performance data.

This data is provided as an illustration only; these fans are not commercially available.

Fan #	Hp	Cubic feet per minute (cfm) at indicated static pressure (inches of water)											
		1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5
Fans 1 through 9 are axial-flow fans													
1	0.33	1,435	620	290									
2	0.5	1,880	960	800	620	380							
3	0.75	1,690	1,460	1,170	780								
4	1.0	2,775	2,500	2,075	1,150	775	500	260					
5	1.5	3,675	3,475	3,275	3,000	2,425	1,700	1,375					
6	3.0	6,400	5,700	5,200	4,500	3,700	2,900	2,200					
7	5.0	9,600	8,550	7,600	6,800	6,150	5,300	4,200	1,550				
8	7.5	13,400	12,500	11,500	10,400	9,000	7,500	6,200	4,450	2,250	1,350	650	
9	10.0	15,700	15,000	14,200	13,400	12,600	11,600	10,500					
Fans 10 through 14 are centrifugal fans													
10	5.0	7,600		6,700		5,800		4,800		3,500		1,500	
11	7.5	9,600		8,900		8,000		7,200		6,100		5,000	
12	10.0	13,450		12,720		11,960		11,120		10,180		9,040	7,450
13	15.0	16,000		15,100		14,200		13,100		11,800		10,000	
14	20.0	21,725		20,430		19,140		17,750		16,140		14,120	11,360

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